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EXPERIMENTAL AND SIMULATED PERFORMANCE OF OPTIMAL CONTROL OF BUILDING THERMAL STORAGE

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ABSTRACT

Dynamic building control involves utilizing the thermal storage potential of building mass to reduce cooling costs. Previous simulation studies have shown that both energy use and peak electrical demand can be significantly reduced using dynamic building control instead of conventional night setback control for many applications. In this study, optimal dynamic building control strategies were compared with night setback control through experiments at a test facility representative of a room in a large office building. Two optimal dynamic building control strategies were considered: minimum total energy costs and minimum peak electrical demand. These control strategies were determined through simulation of the test facility. The experiments showed that up to 51% of the total cooling load could be shifted to off-peak hours through optimal control. The reduction in the peak cooling load was found to be as much as 40%. For the test conditions, estimates of electrical energy and demand savings were determined through simulations and found to be 10% and 38%, respectively. Occupant thermal comfort was also measured during the experiments and maintained within acceptable limits for all control strategies tested. Measured cooling loads compare well with those predicted by the simulation and validate the simulation method. Additional simulations were conducted to study the effect of ambient conditions, utility rate structure, and the coupling between the zone and the ambient through exterior walls on the savings potential of dynamic building control. The results of this research demonstrate the tremendous potential associated with optimal control of building thermal storage.

INTRODUCTION

Conventional cooling control strategies used in large office buildings involve turning cooling equipment off during unoccupied periods. This strategy, often called *night setback control*, minimizes the total building cooling requirement. However, it does not necessarily minimize the utility costs. Dynamic building control utilizes the thermal storage potential of building mass to reduce electric utility costs. An optimal dynamic building control strategy would minimize cooling costs. By precooling the building when it is unoccupied, the thermal storage available in the building mass can be utilized to partially shift cooling loads from daytime to nighttime. Load shifting allows energy costs to be reduced through (1) greater use of lower-cost off-peak electric rates, (2) more favorable ambient operating conditions, (3) operation at more favorable part-load conditions, and (4) the reduction of peak electrical demand.

In many regions of the country, the electric utilities have divided the day into off-peak and on-peak periods. During the on-peak period, which usually runs from late morning to early evening, a higher rate is charged for electricity. Substantial cost savings can be achieved by shifting the cooling load into the off-peak period, even with an increase in total cooling load. In addition, cooling equipment usually operates more efficiently during nighttime and early morning hours because of lower ambient temperatures. When the ambient air temperature is lower than the room temperature setpoint, the use of outside air for free cooling is possible. In addition, many cooling plants operate most efficiently at a fraction of full load. Load shifting can allow operation at more favorable part-load conditions, reducing overall energy use. The final cost-saving benefit of load shifting is the reduction in peak electrical demand. Often utilities will assess a separate charge based on the maximum power drawn during the on-peak hours. Intelligent control of building thermal storage can significantly reduce peak demand charges. Reduction in peak demand would also allow installation of smaller equipment in new construction.

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The potential for thermal storage using the building structure is enormous. A 3-in. concrete slab uniformly subcooled 6°C has 254 $W \cdot h/m^2$ of thermal storage or at least a quarter of a typical office building's 12-hour cooling load of 380 to 900 $W \cdot h/m^2$. In addition to the building structure, a great deal of mass, and therefore storage potential, exists in the office materials such as furniture and paper. Unfortunately, control over the rate of charging and discharging of the thermal storage available using the building structure is limited by comfort considerations and the coupling of the mass to the room air. Therefore, a 6°C temperature swing in the temperature of the building mass may not be attainable in a day.

One of the more comprehensive simulation studies of dynamic building control was conducted by Braun (1990). Braun developed building, cooling system, and weather models and applied dynamic optimization techniques to minimize the total daily cost of operation of a number of systems. A single floor of a large, multistory building with identical floors above and below was modeled. Buildings of both lightweight and heavyweight construction were considered. In addition, several cooling system types, occupancy schedules, weather conditions, and utility rate structures were studied. Savings in total energy use ranged from 0 to 50%, depending on the case examined. When time-of-day electric rates were considered, even greater cost savings were achieved. The primary conclusion was that significant opportunities exist for reducing operating costs in typical commercial buildings through optimal control of building thermal mass.

Other simulation studies have also shown positive results. Snyder and Newell (1990) found cost savings of 18% over night setback control using a dynamic building control strategy applied to the simulation of a lightweight building. Reductions in peak energy use of as much as 50% over night setback were shown in a study by Andresen and Brandemuehl (1992).

Several experimental studies have been conducted (with mixed results) to test the control of building thermal storage. A field experiment was conducted by Ruud et al. (1990) in a building in Jacksonville, Florida. Under a simple precooling strategy, daytime cooling load was reduced but the overall cooling energy required was greatly increased. The building considered was of lightweight construction, and nighttime free cooling could not be employed because of high humidity. The precooling control employed was not optimized but rather provided the maximum precooling possible during the unoccupied times. This control method would maximize load shifting but would not minimize total energy costs.

A second experimental study was performed by Conniff (1991) in a facility designed to represent a room in an office building. Simple alternative control strategies were employed and compared against night setback control in terms of their ability to reduce peak cooling load. The lights in the facility were turned on for a 12-hour period each day. Under night setback control, the room air temperature was maintained at 75°F during the lighting period, and no cooling was provided for the rest of the day. By cooling the room to the daytime setpoint for six hours before the lights came on, the peak cooling load was reduced 28% over night setback. A second precooling strategy involved subcooling the room to 72°F before the lights came on and then maintaining a setpoint at 75°F for the lighting period. This strategy provided no peak reduction when compared to night setback control. Another strategy allowed the temperature to drift upward by 0.1°F at the end of the lighting period and provided an additional 10% peak reduction over the precooling strategy. Conniff suggested that combining the subcooling and temperature drift strategies would lead to an even greater peak load reduction.

One major limitation of previous experimental studies is that the control methods employed in the tests were not well developed. The control strategies were not optimized for the test conditions. Conniff adjusted the control method empirically but did not achieve optimal control. A better approach would be to use computer simulations to determine the optimal control strategy for a specific case. This method was used in this study. The experiments for this study were conducted in the same facility used by Conniff.

The main objective of the research described in this paper was to demonstrate the energy cost and peak demand savings potential of dynamic building control as compared to night setback control for an existing test facility. A second objective was to compare the thermal comfort that would be experienced by the occupants for both types of control. A third objective of this research was to validate the simulation method employed through experimentation so that further simulation studies could be conducted with confidence.

The remainder of this paper is divided into seven sections that present (1) an overview of the approach taken to meet the research objectives, (2) a description of the test facility and instrumentation, (3) a description of the models and important assumptions used in the simulations, (4) an overview of the development of the optimal control strategies, (5) experimental results, (6) estimated cost savings and peak reduction for the experimental data and results from further simulation studies, and (7) conclusions and recommendations.

APPROACH

This research was conducted in two phases: a simulation phase and an experimental phase. A schematic of the approach is shown in Figure 1. In phase one, optimal control strategies were developed through dynamic optimization of a simulation of the test facility. The test facility was modeled from a detailed description of its physical characteristics and operating conditions. A comfort model was used to monitor the simulated occupant thermal comfort conditions and constrain the controlled zone conditions. Models for the cooling system and the weather were also created, allowing the costs associated with any control

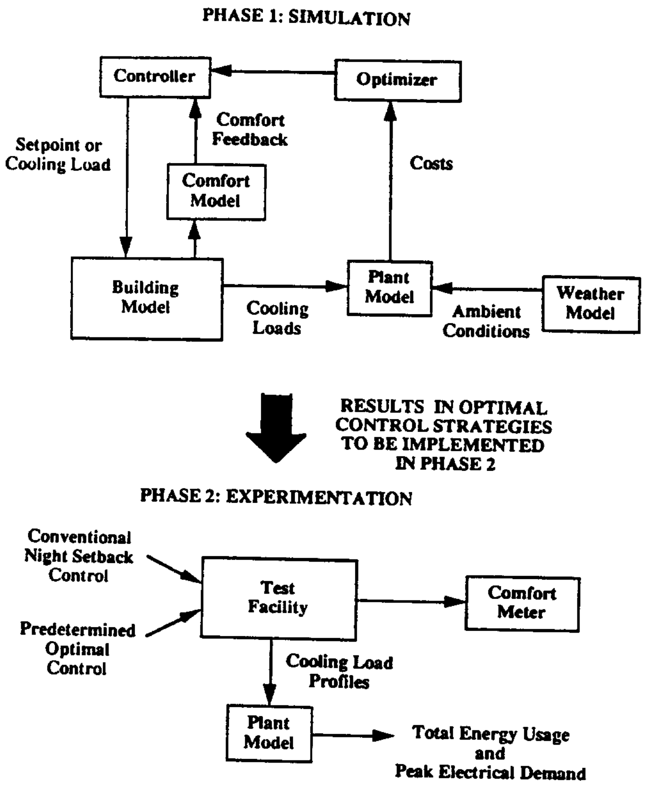

Figure 1 *Approach employed in developing and evaluating optimal control strategies.*

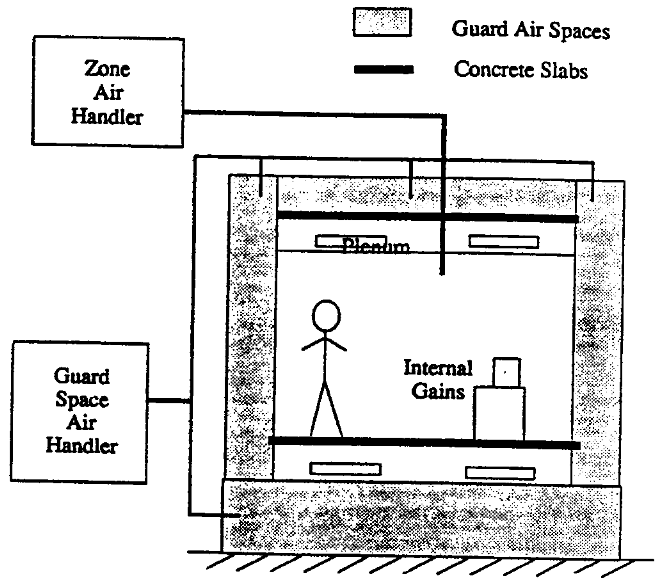

Figure 2 *Test facility.*

strategy to be predicted. The test facility's cooling system was not representative of those installed in typical buildings; therefore, a model of an imaginary representative cooling system was employed. For this test facility, there is no coupling to the ambient, but the weather conditions affect the plant performance and costs. The optimal control strategies were developed for a 24-hour period. Two different optimization problems were addressed: minimum total energy costs and minimum peak electrical demand.

In the second phase, both the optimal control strategies and night setback control were implemented at the test facility. The resulting cooling load profiles under each control were measured and compared. Estimates of energy cost savings and peak electrical demand reduction were determined through simulation. During the experiments, thermal comfort conditions were also measured. The simulated and experimental cooling load profiles were compared to validate the simulation method. Further simulation studies examining other cases were also performed.

THE TEST FACILITY

The facility used in this study was developed to study heating, ventilating, and air-conditioning (HVAC)/lighting interactions (Treado and Bean 1988). The test facility, shown schematically in Figure 2, consists of a 12-ft by 14-ft room with a 2 1/2-in. concrete slab floor, gypsum board walls, a suspended ceiling, and a carpeted floor. A second concrete slab represents the floor above. This construction is relatively lightweight compared with typical commercial buildings. The test room is isolated on all six sides by guard air spaces. These guard air spaces are independently cooled and were controlled to maintain the outside surface temperatures equal to the inside surface temperatures. As a result, the net heat flux into or out of the room is small and the zone can be considered adiabatic. This is representative of an internal zone in a large office building.

Thermopiles with 30 pairs of junctions measure the average temperature difference across each wall, between the tops of the two floor slabs, and between the bottoms of the two slabs. The control system manipulates heating or cooling to each guard air space to keep these thermopile readings at zero.

The room's cooling system consists of a chilled-water coil and a resistance heater. Chilled water is supplied from a central plant. The air handler maintains a constant airflow of 200 cfm, returning air via a ceiling plenum. The air passes over the cooling coil and is reheated to provide the desired cooling load. The supply air enters the room through a ceiling grille. The floor of the test room is elevated to allow for an identical plenum space below. Duplicate lighting fixtures and ductwork are installed in this lower plenum, but the airflow rate of the lower system and the specific fluorescent lights are different.

To simulate an office environment, representative sensible internal gains were added to the facility. A computer (with no monitor), generating 115 W, was placed in the room and left on continuously. Two light fixtures with four fluorescent tubes each, generating approximately 350 W, and a radiator-type electric heater, generating 700 W, were switched on at 6:00 a.m. and off at 6:00 p.m. every day.

This represents a total load of 75 W/m^2 during occupancy and 7 W/m^2 otherwise. Three power transducers measured the electrical power, recording two-minute averages. Moisture gains were not included in this study.

Cooling load supplied to the test facility was calculated from the measured air velocity in the supply duct and the difference between the supply and return air temperatures. The room air temperature was determined by averaging the measurements of 48 thermocouples spaced throughout the room. Six heat flux meters measured the local heat flux through all four walls and the floor and ceiling slabs. To monitor the boundary conditions, the average temperature difference between the inside surface and the outside surface of each wall was measured by 30 point thermopiles. The difference between the temperatures at the top of the floor slab and the top of the ceiling slab, as well as the difference between the bottom of each, was also recorded. These temperature differences provided a measure of the heat fluxes into and out of the test room.

Occupant thermal comfort was represented by Fanger's Predicted Mean Vote (PMV), described in the next section. PMV was determined in two ways. One comfort meter calculated PMV from the dry heat loss of an ellipsoid. Vapor pressure, activity level, and clothing level were input into the device and held constant throughout the tests. A second instrument measured air temperature, two plane radiant temperatures, and air velocity. PMV was then calculated directly from these measured environmental variables. The data presented in this paper were measured by the comfort meter, which employed dry heat loss to calculate PMV. The values calculated from the measured climatic variables were consistently lower by about 0.4 PMV. However, the relative change in PMV over the course of the day was virtually identical for the two instruments. The air temperatures measured by the second instrument were slightly lower than those recorded by the thermocouples in the room, resulting in the lower PMV values.

SIMULATION MODEL

The simulation used to develop and evaluate optimal control for the facility consists of building, cooling system, comfort, and weather models, as depicted in Figure 1. If the control variable is a temperature setpoint, the setpoint is input to the building model, which calculates the cooling load required to meet the setpoint. If the control variable is a constant cooling load, the cooling load is input to the building model and the resulting room air temperature is calculated. The cooling load and the return air temperature, along with the ambient conditions from the weather model, are input to the cooling system model to calculate the total electrical energy required to meet the cooling load. The comfort model computes Fanger's Predicted Mean Vote (PMV) from the thermal conditions calculated by the building model. When determining the cost associated with a control strategy for a 24-hour period, the stored energy in the building at the beginning of the day must equal the stored energy in the building at the end of the day so that the initial energy state of the building does not affect the cost. To ensure that this constraint is met, the models are forced to reach a steady periodic state by applying the same daily control for several identical days.

The building model was extracted from the TRNSYS (Klein et al. 1990) simulation program (subroutine type 56). In this model, walls, floors, and ceilings are represented using transfer function relationships specific to the construction. The model was simplified by neglecting the coupling between the plenum return air and the concrete floor slab. A more accurate model would use two separate zones for the room and the plenum. This simplification is conservative with respect to demonstrating savings for dynamic building control.

The room was assumed to be occupied from 6:00 a.m. to 6:00 p.m. Internal gains were estimated at 15,200 watt-hours (W-h) for a typical office space and added to the model. The gains consisted of a continuous gain of 117 W and an additional gain of 1,033 W during the occupied period. Note that the gains measured in the experiments were slightly higher.

The building model assumes that the air in the room is fully mixed. The inside surface convection coefficients are constant, and the walls are assumed to be adiabatic. A heat balance on the air is used to determine the cooling load required to maintain a setpoint or the resulting air temperature if a specified amount of cooling is provided. It is assumed that there are no moisture gains in the space, so only sensible cooling loads are considered.

The building model requires an estimate of the couplings between the room air, the mass, and the internal gains, which are often difficult to determine accurately. These couplings are defined by the surface convection coefficients and the breakdown of the internal gains between radiation and convection. Convective gains go directly to the air and act as an instantaneous load, while radiative gains are absorbed directly by the surfaces and are only felt as a load after convection to the air. The exact breakdown of internal gains depends on the source of the gains, but a radiative component of [illegible]0% is a reasonable approximation for commercial buildings.

Initially, the surface convection coefficients employed by the model were those recommended in the *ASHRAE Fundamentals* (ASHRAE 1993). However, to improve the accuracy of the model, the convection coefficients for individual surfaces were adjusted using experimental data from the test facility. A step test was conducted in which the lights were turned on at hour zero and the temperature was maintained at 76°F. The simulation convection coefficients were adjusted so that the simulated cooling load profile and surface temperatures gave good agreement with the experimental data. As shown in Table 1, the resulting coefficients for the ceiling and walls are considerably higher than those recommended in the *ASHRAE Fundamentals* (ASHRAE 1993). In recent work by Spitler et al. (1991),

TABLE 1
Comparison of Surface Convection Coefficients (Btu/h·ft^2)

Surface	Model Values	ASHRAE Values	Spitler's Values
Ceiling	1.4	0.17	3.0
Walls	1.1	0.54	1.1
Floor	0.3	0.71	0.8

surface convection coefficients were measured experimentally and correlated to the jet momentum number. The jet momentum number for the test facility was not known, but the airflow was 9 air changes per hour (ACH) when the fans were on. Convection coefficients found by Spitler for a room with a ceiling inlet and 15 ACH are also shown in Table 1. The convection coefficients for the ceiling and walls determined from the test data are between those recommended by ASHRAE and those found by Spitler. The floor convection coefficient is considerably lower than the ASHRAE and Spitler values. This most likely is due to the insulating effect of the carpeting.

The cooling system model consisted of two parts: the cooling plant (chillers, cooling towers, and pumps) and the air handlers. The model assumed that the equipment was always operating at steady state to meet the requirement for each simulation time step. An empirical model previously developed by Braun (1988) was used to relate plant power consumption to two variables: total chilled-water load and the difference between the supply air temperature and the ambient wet-bulb temperature. The plant considered in this study used a variable-speed chiller, variable-speed cooling tower fans, and variable-speed pumps. This plant had favorable part-load operating characteristics and was termed a "good" plant in Braun's study. The air handlers were assumed to have variable-speed drives, with a cubic relationship between power consumption and supply airflow rate.

A dry-bulb economizer was used to control free cooling available from outside air in the simulation. If the ambient air temperature was lower than the return air temperature, outside air was brought in and the return air was exhausted. Although there were no restrictions on free cooling at high ambient humidities, this would be needed in a practical application. In addition, cooling requirements due to moisture removal were not considered. Since there are no internal moisture gains, this assumption is valid for dry ambient conditions. For high ambient humidities, dehumidification would increase the total cooling requirements.

Fanger (1970) developed a model of human thermal comfort in terms of measurable environmental factors. He devised a comfort scale termed the *Predicted Mean Vote* (PMV), on which a value of 0 represents ideal thermal comfort for the average person, negative values denote cool conditions, and positive values represent a warm environment. Table 2 shows the relationship between the PMV scale and the average person's perception of thermal comfort. In general, a PMV of ± 0.5 falls well within acceptable comfort limits as specified by ASHRAE.

TABLE 2
Relationship Between PMV and the Average Person's Perception of Thermal Comfort

PMV	Thermal Comfort
3	Hot
2	Warm
1	Slightly Warm
0	Neutral
−1	Slightly Cool
−2	Cool
−3	Cold

A comfort model was developed using Fanger's equations to calculate PMV from the air temperature and the surface temperatures estimated with the building model and assuming constant values for activity level, clothing, and humidity. Air velocities were assumed to be low enough that natural convection would dominate. Light office activity (60 kcal/h·m^2), clothing consisting of slacks and a long-sleeved shirt (f_{cl} of 0.6 and I_{cl} of 1.19), and a vapor pressure of 10 mm Hg were used. The mean radiant temperature was calculated from an angle-factor weighting of the simulated surface temperatures as described in Fanger (1970). A PMV of ± 0.5 was assumed acceptable for all simulations. The primary purpose in simulating comfort was to compare conditions for the alternative control strategies. Comfort was also used as a constraint in the determination of the optimal control strategies described in the next section.

To determine the total energy use of the cooling system, hourly weather data consisting of the ambient dry-bulb and wet-bulb temperatures were required. To eliminate the effects of random weather patterns, diurnal temperature variations were generated from statistical correlations. Correlations relating the diurnal variations of dry- and wet-bulb temperatures to the average daily dry-bulb temperature and the clearness index were developed by Erbs (1984) and were used for the weather model.

OPTIMAL CONTROL STRATEGIES

The optimal control determined for any 24-hour period is a function of the building's characteristics, the cooling system's characteristics, and the ambient conditions. The optimal strategies are specific to the case for which they are developed: a strategy that is optimal for one set of weather conditions is not optimal for another set of conditions.

Optimal control strategies were developed for two different cases: minimum daily energy costs and minimum daily peak electrical demand. In a real application of dynamic building control, the controller would try to minimize total energy costs during most of the year. For the design day (and days close to design), reduction of peak demand would become more important than minimizing energy costs. Peak demand can often be reduced at the expense of higher total energy costs. The ideal trade-off depends on the utility rate structure. The goal of this research was to show the maximum potential for both energy savings and peak power reduction.

For the energy minimization control, a day with an average ambient temperature of 80°F was selected. The maximum temperature was 90°F, the minimum temperature was 72°F, and the sky was assumed to be partly cloudy with a solar clearness index of 0.6. On this day, there was little opportunity for free cooling, so energy savings were mainly due to better efficiency of the mechanical cooling equipment at night. The peak minimization day selected was for an average ambient temperature of 75°F. The maximum temperature was 85°F, the minimum temperature was 67°F, and again the sky was partly cloudy. For both of these strategies, the plant considered had "good" part-load characteristics, and no time-of-day rates were assumed.

The optimization problem was to determine the values of the hourly zone temperature setpoints that minimized a specific cost function. The energy minimization cost function was

$$\text{Cost} = \sum_{k=1}^{24} P(T_{z,k} \text{ or } Q_{z,k}) \times R(k) \times \Delta t, \qquad (1)$$

where

k = hour,
P = hourly electric energy rate to obtain an air temperature setpoint ($T_{z,k}$) or an hourly cooling load ($Q_{z,k}$),
$R(k)$ = cost per unit energy, and
Δt = time interval (one hour for this case).

The peak minimization cost function was

$$\text{Peak} = \max(P_j + B_j, P_{j+1} + B_{j+1}, \ldots, P_{j+n-1} + B_{j+n-1}), \qquad (2)$$

where

B = noncooling building power,
j = first hour of the on-peak period, and
n = number of hours in the on-peak period.

Note that B is constant during the on-peak time for the conditions of this study. For each hour, the control variable used in the cost functions was either the air temperature setpoint or the cooling provided to the building. The optimization was constrained by the condition that thermal comfort be maintained during all occupied times, such that $-0.5 \leq \text{PMV} \leq 0.5$.

Neither of the cost functions given in Equations 1 and 2 is a smooth function of the control variables due to the different possible modes of operation of the cooling equipment. To solve these optimization problems, the direct search complex method was employed. The complex method only employs functional comparisons and does not require derivative information. A description of this method can be found in Rao (1984).

Braun (1990) showed that for energy minimization control, the optimal temperature setpoints during occupancy should always be set at the highest allowable temperature within the comfort range. His results were for buildings that were coupled to the ambient. This strategy ensures the quickest discharge of the mass, giving the best storage efficiency. With time-of-day rates, this step up to the maximum temperature would occur at the onset of the on-peak period. In this study, the zone was not coupled to the ambient, so Braun's strategy may not be the true optimal for the facility. However, the occupied setpoints were set at 76°F (the assumed upper comfort limit) so that a fair comparison of thermal comfort between the energy minimization control and night setback control could be made. The optimal control problem is then reduced to determining the unoccupied setpoints.

To reduce the computation time required to solve the optimization problem, simplifications in the control strategy were identified. Initial optimization results revealed that the optimal cooling load was fairly constant during unoccupied times. Using these results, the number of control variables was reduced to three—two constant cooling loads for the unoccupied period and the time associated with changeover between the two. This method reduces the computation time while giving results close to optimum. Several cases were studied and found to be within 2% of the results obtained using 12 unoccupied period control variables.

The predicted cooling loads and air temperatures under optimal control for an average ambient temperature of 80°F are shown in Figures 3 and 4, respectively. The optimal control strategy calls for considerable precooling of the room, lowering the room air temperature to 68°F. From 6:00 p.m. to 5:00 a.m., a constant cooling load of 520 W is supplied. At 5:00 a.m., the cooling load is reduced to 476 W. After 6:00 a.m., no cooling is provided until the room air temperature reaches 76°F. At this time, the cooling system comes back on and maintains a room air

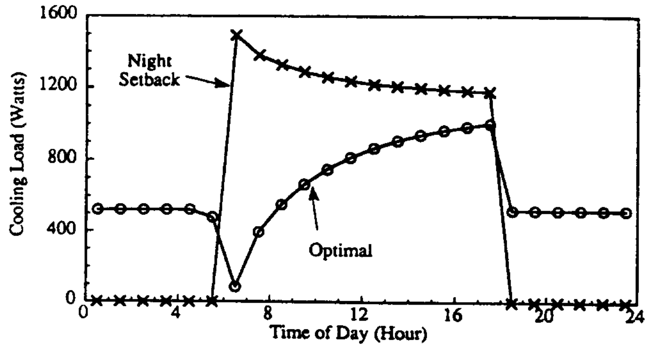

Figure 3 *Simulated cooling load profiles for energy minimization control and night setback control with an average ambient temperature of 80°F.*

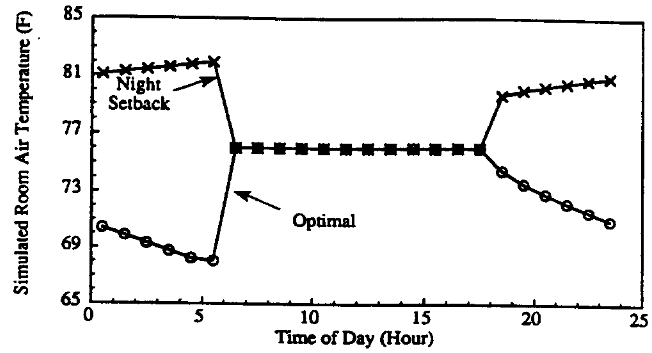

Figure 4 *Simulated room air temperatures for energy minimization control and night setback control with an average ambient temperature of 80°F.*

temperature of 76°F until 6:00 p.m. The simulation predicts that 1.76 ton-hours, or 41% of the daily cooling load, will be shifted into the nighttime hours. The simulation estimates a savings of 15% in total energy cost for optimal as compared with night setback control.

For the peak minimization problem, the three-variable optimization was also used to determine the control for the unoccupied period. The on-peak period was assumed to be from 6:00 a.m. to 6:00 p.m. To obtain the maximum peak reduction, the occupied period should begin with the thermal comfort at the lower bound of the acceptable comfort range and the occupied setpoints should be manipulated to control the discharge of the thermal storage in the mass.

Figure 5 is a plot of the cooling load for the peak minimization control with an average ambient temperature of 75°F. The corresponding room air temperature is plotted in Figure 6. From 6:00 p.m. to 3:00 a.m., a constant cooling load of 967 W is supplied to the zone. During this time, the air temperature is cooled to about 60°F. At 3:00 a.m., the cooling equipment is shut down so that the air temperature can recover to meet the comfort constraint. During the on-peak period, the air temperature setpoint is adjusted upward in order to maintain a constant power, resulting in a relatively constant cooling rate. The simulation predicts a peak cooling load reduction of 852 W (2,910 Btu/h) or 57%, which results in a 61% reduction in peak electrical demand due to cooling. If the on-peak period did not begin until later in the day, the optimal control would be virtually the same, but the peak reduction could be lower. The peak electrical demand under night setback control occurs in the morning because there is no ambient coupling. The potential for reducing afternoon peaks with optimal control is greater for ambient-coupled buildings, in which peak demand typically occurs in the afternoon.

In the peak minimization experiments described in the next section, either temperature setpoint control (Figure 6) or cooling load control (Figure 5) was used during the on-peak period. For both control methods, a constant cooling load of 967 W is supplied to the zone from 6:00 p.m. to

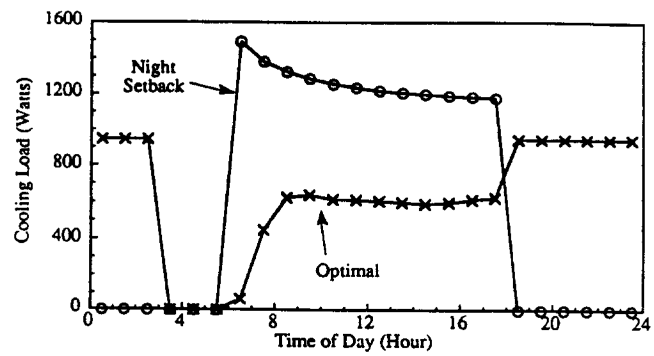

Figure 5 *Simulated cooling load profiles for peak demand minimization control and night setback control with an average ambient temperature of 75°F.*

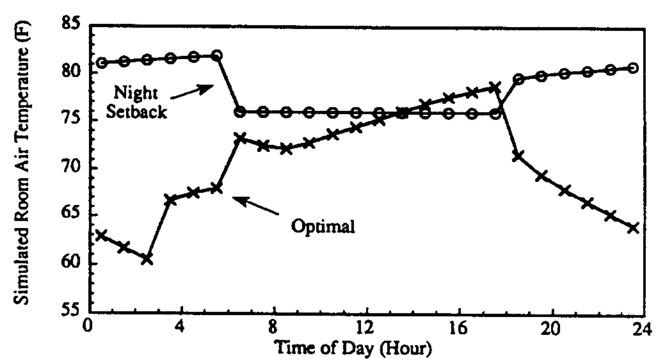

Figure 6 *Simulated room air temperatures for peak demand minimization control and night setback control with an average ambient temperature of 75°F.*

3:00 a.m. Under temperature control, when the room air temperature reached 73°F, the cooling equipment was turned on and controlled the temperature setpoints defined by the following formula:

$$\begin{aligned}\text{Setpoint} &= 73.5 - 0.5 \times (\text{time} - 6) \quad \text{when } 6 < \text{time} < 9 \\ &= 72 + 0.778 \times (\text{time} - 9) \quad \text{when } 9 < \text{time} < 18.\end{aligned} \tag{3}$$

With cooling load control, the cooling equipment was turned on at 6:00 a.m. and the cooling load setpoints were

$$\begin{aligned} Cool &= 352 \times (time - 6) \\ &\quad \text{when } 6 < time < 8 \\ &= 704 \text{ when } 8 < time < 18. \end{aligned} \tag{4}$$

If the room air temperature went over 79°F, the cooling load was increased to maintain the 79°F setpoint. The cooling loads specified for the experiment are slightly higher than those predicted by the simulation (Figure 5) because the internal gains in the experiment were higher than those used in the simulation.

EXPERIMENTAL RESULTS

The main goal of the experiments was to demonstrate the load-shifting potential of the building mass. The case examined was not the ideal application of dynamic building control in that the building is of lightweight construction, no office materials were present, and the floor was carpeted. If significant load shifting could be achieved in this case, the potential exists in many buildings. The second objective of these experiments was to validate the simulation method and verify the predicted results. This would allow the use of the simulation to study conditions beyond those considered in the tests. Finally, thermal comfort measurements from these experiments were used to compare thermal comfort under dynamic building control and night setback control.

Three separate sets of control tests were performed: (1) night setback, (2) energy minimization, and (3) peak minimization. The cooling system serving the test facility was controlled to maintain either specified temperature or cooling supply setpoints, depending on the predetermined control strategy. Each test was performed for a minimum of three days to allow a steady-periodic condition to be obtained. The effect of the initial thermal state of the test facility must be removed in order to have meaningful comparisons between alternative control methods. Figure 7 is a graph of the cooling loads for a night setback test conducted over a five-day period. The cooling load on the first day is considerably lower than the steady-periodic case because the test facility was cool at the beginning of the test. By the fourth day, this initial condition no longer affects the cooling load profile.

For the night setback test, the cooling system was turned on at 6:00 a.m. with a room temperature setpoint that was maintained at 76°F until 6:00 p.m. At this time, the cooling system (including the fans) was shut down and the temperature was allowed to drift until 6:00 a.m. the next day. For the energy minimization test, the optimal control determined by the simulation and described previously was employed.

The resulting cooling load profiles for the night setback and energy minimization tests are very different, as can be seen in Figure 8. The night setback test has a considerable spike at 6:00 a.m. Some of the radiative gains from the

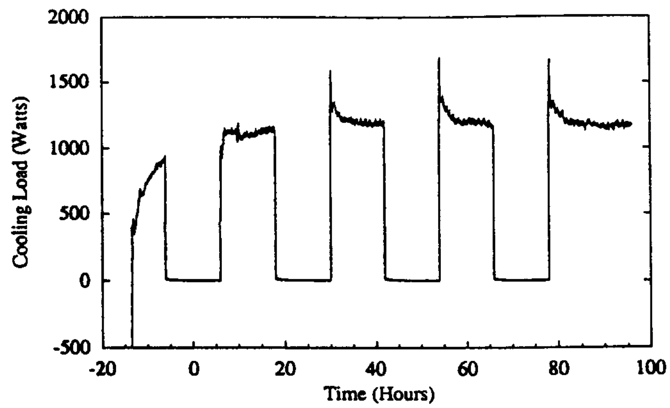

Figure 7 *Measured cooling loads for night setback test.*

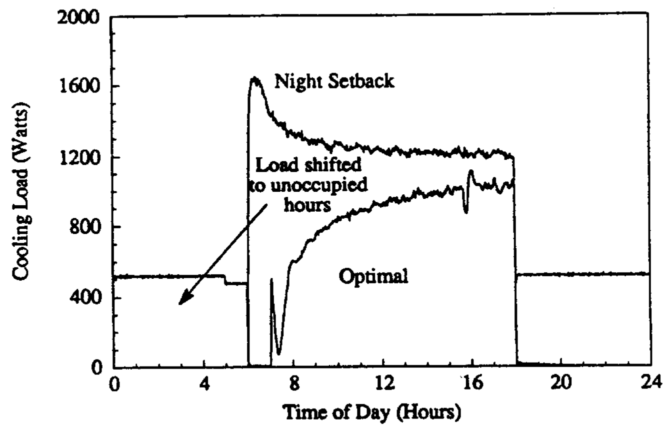

Figure 8 *Comparison of measured cooling load profiles for energy minimization control and night setback control.*

previous day, as well as small nighttime gains, caused the air and mass temperature of the facility to rise during the night, leading to a high initial load in the morning. The cooling load for the energy minimization test increases throughout the occupied period as the building structure is heated.

The daytime cooling load for the energy minimization test is much less than for the night setback test, with 5,700 W-h (1.6 ton-hours) of cooling having been shifted into the nighttime hours. This shift is approximately 37% of the total daily cooling load. The spike in cooling load for the energy minimization test occurring at 6:00 a.m. was due to overshoot by the controller. Another controller problem occurred at 4:00 p.m. due to a change in the chilled-water supply temperature. In addition to the significant load shifting, the peak cooling load of the energy minimization test is also less than that of the night setback test. The amount of peak demand reduction would depend on the time covered by the on-peak period. For a peak period from 6 a.m. to 6 p.m., the peak cooling load under energy minimization control was 1,068 W, or about 35% less than the peak under night setback control.

The thermal comfort maintained by the two control strategies is plotted in Figure 9. In the night setback test, the space is uncomfortably warm during the unoccupied period but is brought down to the upper comfort limit by 6:05 a.m. The room is uncomfortably cool at night for the energy minimization test, but the temperature quickly recovers when the cooling equipment is turned off at 6:00 a.m. Lights and other internal gains were turned on at 6:00 a.m. and acted to heat the space. By 6:30 a.m., comfort is acceptable, and from 7:45 a.m. to 6:00 p.m., comfort is nearly identical to that maintained under night setback control. In order to improve comfort between 6:00 a.m. and 6:30 a.m., the lights could turn on earlier, the amount of precooling could be reduced, or the zone-heating system

As described in the previous section, peak minimization control determined through simulation was implemented in two different ways in the test facility. One method employed temperature setpoint control during the occupied period to level cooling load, while the other method controlled cooling load directly during the occupied period. The resulting strategies are defined by Equations 3 and 4. If the simulation had been perfect, the two methods would have given identical cooling load profiles, but there were differences. The advantage of direct load control is that it could guarantee maximum peak reduction in the presence of unexpected internal gains. However, it does not guarantee that comfort would be maintained. During the load control test, the control method reverted to temperature control if the room air temperature exceeded 79°F.

The cooling load profiles for the two peak minimization tests are compared with the night setback test results in Figure 10. The temperature control method achieved a 40% reduction in peak afternoon cooling load. However, the cooling load was higher between 8 a.m. and 10 a.m. than that predicted in simulation. There are two reasons for the differences. First of all, the controller did not precisely maintain the desired cooling load at night, due to an error in the control program. This error was corrected for the load control test. Second, some unmeasured heat gains may have entered the space from the boundaries during the tests, since the total cooling load measured was found to be greater than the internal gains.

The cooling load control test did not maintain a constant load in the afternoon because the room air temperature reached 79°F prior to the end of the occupied period. The total cooling load supplied in this test was higher than the internal gains, again probably due to unmeasured heat gains entering the space through the boundaries. Even with these problems, both control methods were successful in shifting about 8,000 W-h (2.3 ton-hours) of cooling into the nighttime period. This represents 51% of the total cooling load. The peak cooling load under night setback control was 1,651 W. The peak was reduced to 938 W and 935 W under the temperature and load control methods, respectively. This is a reduction of 43% in peak cooling load and is significantly larger than the 35% reduction under energy minimization control.

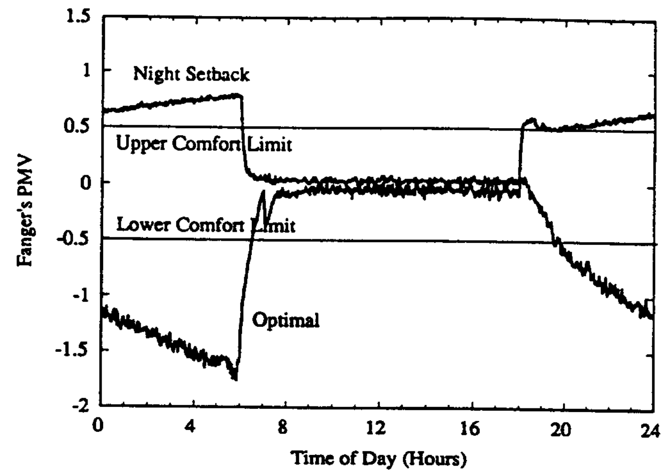

Figure 9 *Comparison of Fanger's PMV under energy minimization control and night setback control.*

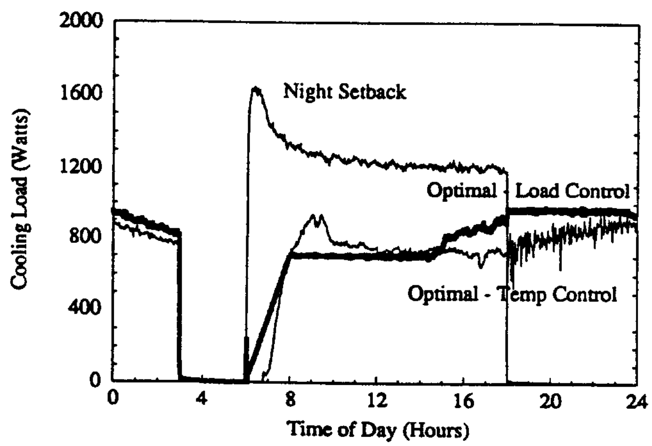

Figure 10 *Comparison of measured cooling load profiles for peak demand minimization control and night setback control.*

If the time of the on-peak period were changed, the optimal control strategy developed through simulation would be different. However, examining the results of these experiments gives an idea of the potential of optimal control in reducing peak cooling load for other on-peak periods. For an on-peak period from 10:00 a.m. to 6:00 p.m., peak cooling load was reduced 530 W and 340 W for the temperature and load control methods, respectively. These reductions are significantly larger than the 180-W reduction achieved under energy minimization control. Furthermore, afternoon peak reductions could be greater for a typical building with ambient coupling. In this case, the peak cooling rate for night setback control would occur in the afternoon.

The thermal comfort maintained during the peak minimization test using the load control method is plotted in Figure 11. The air temperature was cooled to 58°F by 3:00 a.m. but recovered to 68°F by 6:00 a.m. PMV reached an

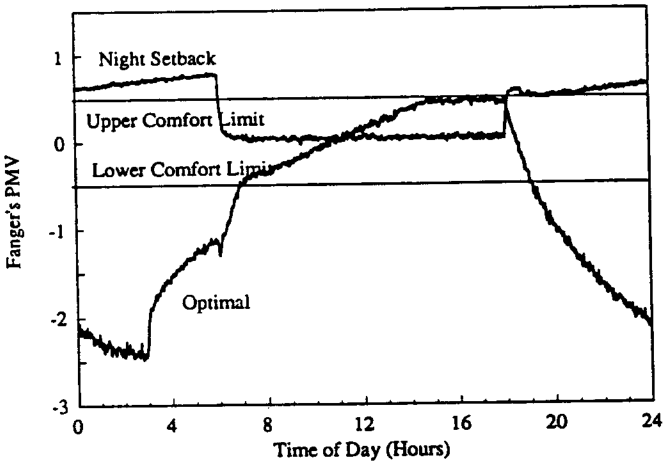

Figure 11 *Comparison of Fanger's PMV under peak demand minimization control and night setback control.*

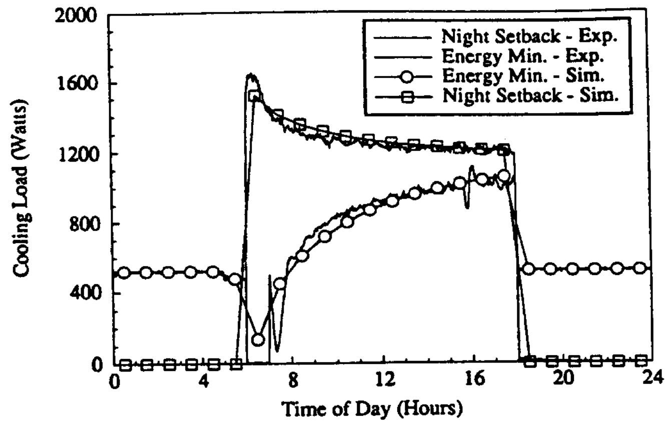

Figure 12 *Comparison of measured and simulated cooling load profiles for energy minimization control and night setback control.*

acceptable level at about 7:00 a.m. Since this control method represents a limiting case that would only be used once or twice a year on a design day, the one hour of minor discomfort in the morning would most likely be acceptable. After 7:00 a.m., thermal comfort gradually increased until 3:00 p.m., when the upper temperature constraint of 79°F was reached. Comfort data for the temperature control test were not recorded properly and are not available. It is reasonable to assume that the comfort results would be similar for both tests.

The uncertainty in any one measured variable at the test facility is generally less than 3%. A second night setback test was conducted to verify repeatability of the facility. The measured cooling loads were within 4% of those found in the first test. The biggest source of error is unmeasured heat flux through the boundaries of the test facility. For the night setback and energy minimization tests, the total daily cooling provided was within 2% of the measured internal gains. However, the total cooling provided for the peak minimization tests was 6% greater than the internal gains. The larger error for this test was probably due to the lower room air temperatures that were maintained during precooling. Although the guard air spaces maintained the average outside surface temperatures equal to the average inside surface temperatures, heat gains still entered the facility due to flux at the edges of each wall. These heat gains are greater at lower room air temperatures.

After the tests were completed, the measured internal gains were used in the simulation to compare the predicted and test results. As shown in Figure 12, both the simulated night setback and energy minimization control cooling loads agree well with the measured cooling loads. The simulation predicts that the energy minimization control will allow 5,912 W-h (1.68 ton-hours) of cooling to be shifted into the unoccupied hours. In the experiments, the measured load shift was 5,760 W-h (1.64 ton-hours). The simulated load matches the experimental load exactly at night because the facility was controlled based upon cooling load.

The actual and predicted room air temperatures for both the night setback and energy minimization tests are shown in Figure 13. The night setback simulation results agree well with the measured values. However, the temperatures predicted by the energy minimization simulation are considerably higher than those measured. This discrepancy caused an overestimation of the thermal comfort at 6:00 a.m. In developing the simulation method, a number of assumptions were made. The building model was tuned using test data for one specific set of conditions. When there is no airflow or the lights are not on, the accuracy of the model is reduced. Since the heat capacity of the air is low, errors in the model amplify errors in predicted air temperatures. These errors do not have a large effect on the predicted cooling load profile but do affect the modeled thermal comfort.

SIMULATED ENERGY SAVINGS AND PEAK DEMAND REDUCTIONS

The experimental data were the basis for comparing the cooling load profiles for each type of control. However, it is also important to estimate the potential for energy cost savings and peak electrical demand reduction. The test facility's cooling system is not representative of those installed in typical buildings. To obtain the total energy required to meet the cooling loads, a model of a representative cooling system was employed. When determining the costs in this manner, the idealizations and assumptions of the cooling system model must be considered. Actual cost data would be better and tests obtaining such data should be conducted.

Using the cooling system model to determine the costs associated with each control strategy, the energy minimization control strategy reduced total energy costs by 10% over night setback control for the test day chosen. This is lower than the 15% predicted because the internal gains in the test facility were 3% higher than those used in the simulation to develop the control strategy.

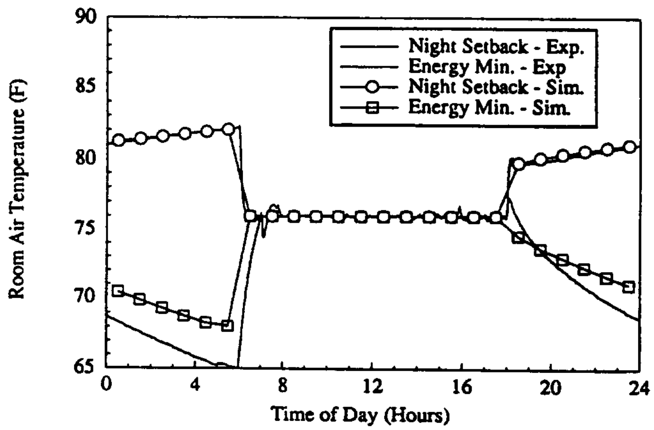

Figure 13 *Comparison of measured and simulated room air temperatures for energy minimization control and night setback control.*

In determining the control strategy for peak minimization in simulation, the on-peak period was 6:00 a.m. to 6:00 p.m. The peak electrical demand due to cooling was reduced 38% for both the temperature control and load control peak minimization tests when compared to the peak electrical demand due to cooling under night setback control. If the on-peak period were from 10:00 a.m. to 10:00 p.m., the peak electrical demand due to cooling would be reduced 51% for the temperature control peak minimization test. This is because the peak electrical demand under night setback control occurs at about 10:00 a.m.

The potential savings for dynamic building control are affected by the ambient conditions, the utility rate structure, and the coupling between the zone and the ambient through external walls. These effects were studied for the facility through simulations. Figure 14 shows the percent savings of energy minimization control over night setback control as a function of average daily ambient temperature for four cases. The standard case represents the one used in the tests. The percent savings increase with decreasing ambient temperature since free cooling is more effectively utilized by the energy minimization control. The magnitude of the energy savings is quite substantial at high ambient temperatures. If the on-peak charge for energy is twice the off-peak charge (2-1 time-of-day rates) with an on-peak period from 6:00 a.m. to 6:00 p.m., the percent savings over night setback are significantly higher. This is especially true at higher average ambient temperatures, when free cooling is not available and most precooling must be done mechanically. For the design day (average ambient temperature of 85°F), the percent of savings over night setback increase from 5% to 12%.

The conditions for the test zone did not include coupling to the ambient. The effect of ambient coupling on the control comparisons was studied through simulation. The south wall in the test facility model was changed to an 8-in. cinder-block wall with insulation-filled cores and 40% of

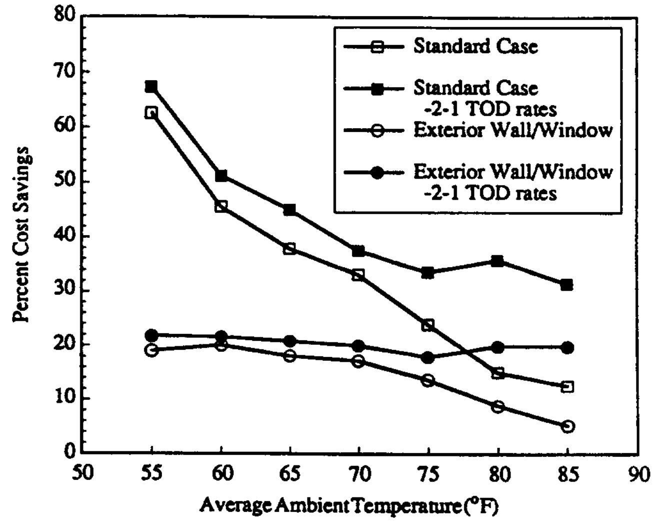

Figure 14 *Simulated percent cost savings of energy minimization control over night setback control.*

the wall area a double-glazed window. Figure 15 is a plot of the cooling load profiles for both night setback control and the energy minimization control for a daily average ambient temperature of 80°F. With one of the walls replaced with an exterior wall having a window, the cooling load profiles change dramatically. The peak cooling load occurs at about 1:00 p.m. for both night setback and energy minimization control because of heat gains due to convection with the warmer ambient air and solar radiation. These results are more typical of the cooling load profiles for commercial buildings. Compared with the previous results, the percent of load shifted to unoccupied hours through optimal control is reduced (25% vs. 41%), but the amount of load shifted is almost the same (1.71 ton-hours vs. 1.76 ton-hours).

With the exterior wall installed, the percent savings associated with energy minimization control compared to night setback control are shown in Figure 14. The percent savings are less than for the standard case at lower ambient temperatures. The coupling with the outside air helps to cool the space at night, even under night setback control. As the average ambient temperature increases, the percent cost savings decline. The increase in envelope losses due to precooling begins to outweigh the gains due to more efficient equipment operation. Eventually, at very high ambient temperatures, the optimal control strategy would be night setback control. The simulated percent savings obtained with the 2-1 time-of-day rates described previously with the exterior wall are also shown in Figure 14. As with the standard case, the percent savings are considerably higher than the case with no time-of-day rates at high ambient temperatures.

The simulated peak electrical demand for both night setback control and peak minimization control is shown in Figure 16. The percent reduction in peak electrical demand

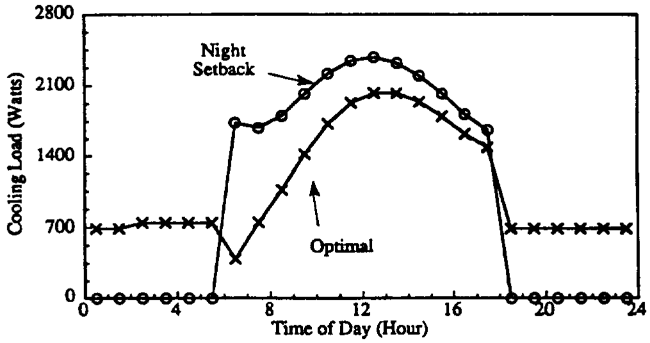

Figure 15 *Simulated cooling load profiles for energy minimization control and night setback control with an average ambient temperature of 80°F and an exterior wall with a window.*

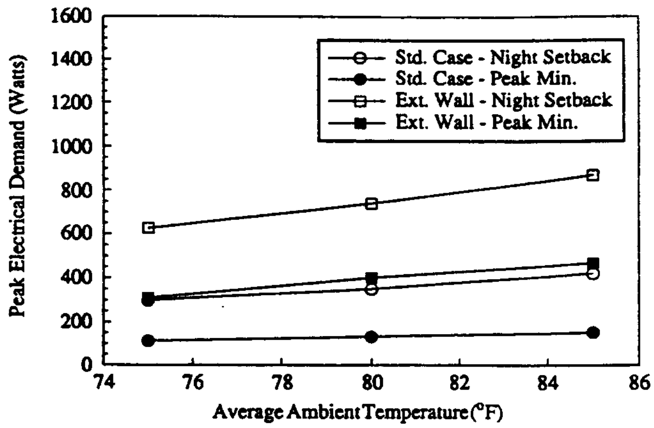

Figure 16 *Peak electrical demand for both night setback control and peak minimization control with and without exterior wall with window.*

due to cooling is approximately 62% for the standard case. Also shown in Figure 16 are results for a room with an exterior wall and window as described previously. The peak electrical demand due to cooling is much greater because of the increased load due to the coupling with the ambient. The percent peak reduction has decreased to 50%, but the magnitude of the reduction is greater. In addition, the maximum peak reduction occurs in the afternoon, which is more typically within the period for on-peak electrical rates.

CONCLUSIONS

Previous simulation studies have shown significant reductions in energy costs and peak electrical demand for dynamic building control over night setback control. In this study, these results were validated through experiments. Simulation methods were used to determine the optimal control strategies for a test facility to minimize total energy and to minimize peak electrical demand. Both optimal control strategies and night setback control were implemented at the test facility. The cooling loads and thermal conditions were measured for each control, leading to the following results and conclusions:

1. The thermal storage available within a building's structure is quite large, even with the lightweight construction and carpeted floor used in this study. Up to 51% of the total cooling load—2.2 ton-hours—was shifted from daytime to nighttime hours for the test facility.
2. Compared with conventional night setback control, the peak cooling load was reduced 15% through energy minimization control and 40% through peak minimization control.
3. Acceptable thermal comfort was maintained by both dynamic building control strategies during occupied hours. Between the hours of 6:00 a.m. and 6:00 p.m., the conditions were outside the comfort range for only 30 minutes in the energy minimization test and for one hour in the peak minimization test. Better comfort control is possible by limiting the nighttime precooling or using the heating system just prior to occupancy; however, this would reduce the savings.
4. The simulation accurately predicted the cooling loads. Predicted air temperatures were slightly higher than experimental values.

A model of a representative cooling system was used to estimate energy costs associated with each control strategy for the test conditions. The effects of weather conditions, utility rate structures, and coupling with the ambient were also examined through simulation. From these simulations, additional conclusions were drawn.

5. For the test conditions, energy minimization control achieved a 10% reduction in total energy use over night setback for the simulated cooling system without time-of-day rates.
6. For the test results, the simulated peak electrical demand due to cooling would be reduced 38% for optimal control as compared with night setback control.
7. The percent savings for optimal control increase significantly at lower average ambient temperatures due to greater nighttime free cooling.
8. When 2-1 time-of-day electric rates were employed, the percent of cost savings was significantly higher, especially at higher ambient temperatures, where precooling required the use of mechanical cooling equipment. With a daily average ambient temperature of 80°F, energy costs for cooling were reduced 36%.
9. When one wall of the test facility was coupled to the ambient, the percent of cost savings dropped significantly, but the magnitude of savings was almost identical. The percent peak electrical demand reduction was also lower for the facility with an external wall, but the magnitude of the reduction was greater.

These results show the value of determining the optimal control method for a specific application through simulation. These experiments were conducted in the same facility in which Conniff (1991) performed his work. However, his results were not as encouraging as these because the control method was not optimized.

The control strategies developed in this study are only optimal for this specific test facility. The building construction, occupancy schedule, cooling system, and simulated weather conditions were all important in developing the control strategies. It would be possible to develop a rule-based controller that employed measured values such as ambient temperature and zone air temperatures to determine the optimal control for this facility. However, this controller would probably not work well for another application. Because of the number of application-specific variables, an adaptive controller that learns the characteristics of the system over time is probably the best approach.

It is important to note that the test facility and its use in this study were not necessarily representative of an actual building. In particular, (1) the structure is relatively lightweight in comparison with typical commercial buildings, and there were no office furnishings or materials present; (2) the facility was operated as an internal zone with no coupling to the ambient; and (3) the energy loads were all sensible (i.e., no dehumidification). These characteristics have different effects on the savings potential associated with using building thermal storage. The lightweight structure and lack of furnishings yield energy and demand cost savings that could be very conservative. The effect of adding ambient coupling is to reduce energy savings but increase the potential for reducing afternoon peak demand. The afternoon peak savings are increased with ambient coupling because the peak demand for conventional control is shifted from morning to afternoon. The assumption of no latent cooling requirements should affect both conventional and optimal control in a similar fashion, unless the ambient humidities are high enough to restrict the use of nighttime ventilation cooling.

Primarily, the experiments were meant to demonstrate that the savings predicted through simulation could be achieved in a real structure. Simulations were then used to study the effects of ambient coupling and other changes from the test conditions. Furthermore, Braun (1990) used the same simulation approach to show significant savings for a range of building, plant, utility rate, and weather characteristics. This study validates those results and further demonstrates the tremendous potential associated with optimal control of building thermal storage.

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